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(54) **Brush seal**

(57) A brush seal seals chambers, at different pressures, at a circumferential gap between a machine housing 1 and the shaft 2. The seal consists of a bristle bundle B mounted in a holder 3 from which it is guided radially inwardly between circumferential flanges 4,5 in order to seal against the shaft. In order to protect the bristles from turbulent flow a cover ring 6 for the bristle bundle is arranged radially and axially movably in the gap between the bristles and the flange 5 on the high-pressure side. The differential pressure urges the cover ring against the bristle bundle. The cover ring projects beyond the inner edges of the flanges and encloses the machine shaft, and in extreme eccentric positions of the shaft (Fig.3b) reduces the maximum gap to twice the clearance of the ring from the shaft.

FIG.1

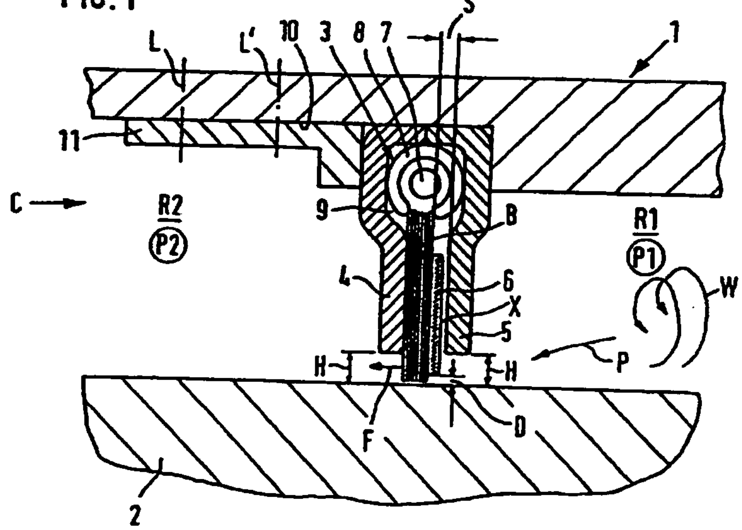

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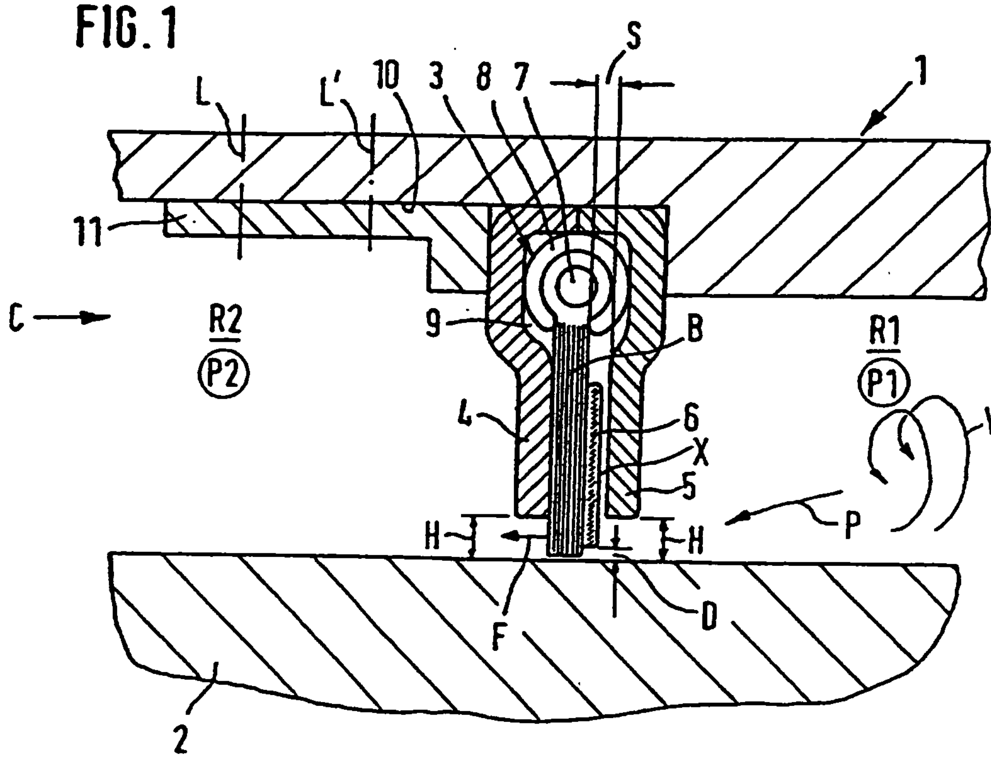
FIG. 1



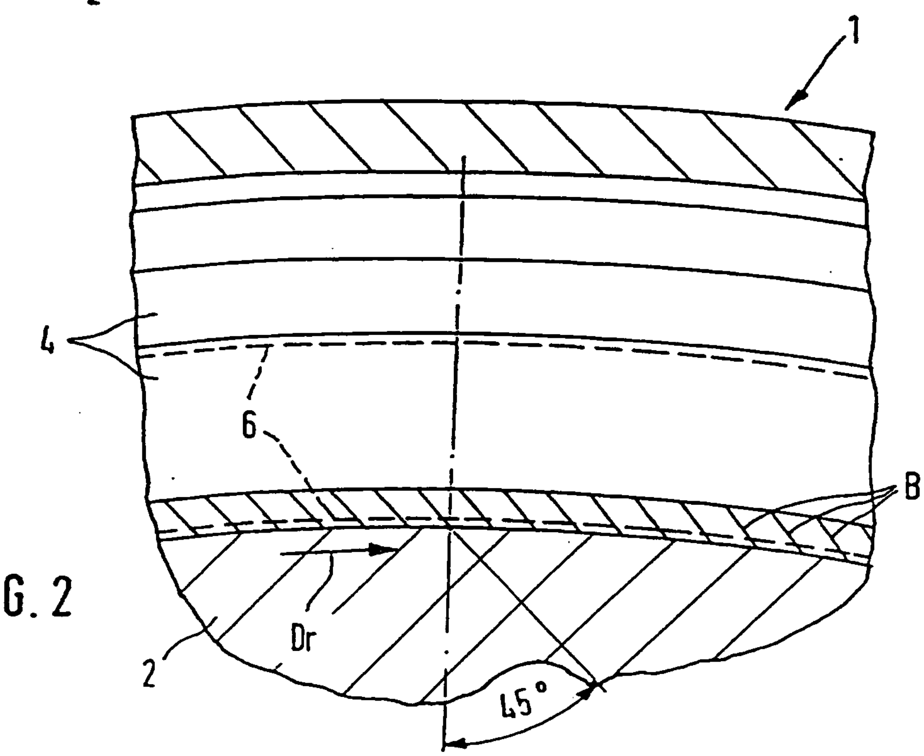
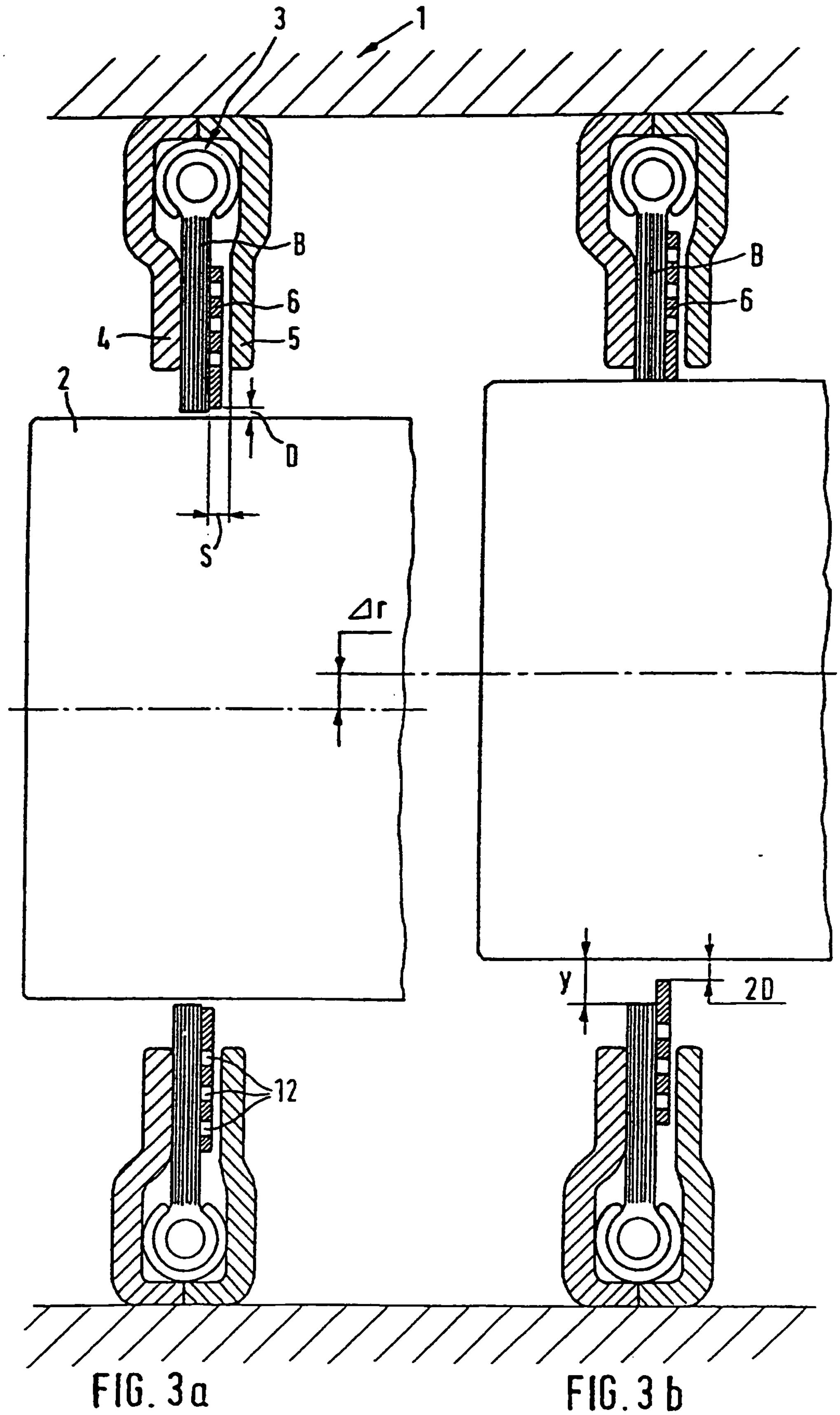


FIG. 2


1
3
B
6
4
5
2
D
S
Δr
12
FIG. 3a
B
6
y
2D
FIG. 3b

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BRUSH SEAL

The invention relates to a brush seal, particularly for use in turbine engines.

Brush seals are used in turbo machines, particularly gas turbine engines, to seal chambers containing different fluid pressures from one another at ring-shaped gaps, e.g. between a machine housing and a machine shaft, with the smallest possible leakage flow. Such seals are disclosed, for example, in DE 39 07 614 A1. By using these seals, shaft eccentricities resulting from rotor imbalances should be compensated for by a bristle movement. However, significant shaft eccentricities are often linked to a high leakage flow at the seal.

It is a fundamental problem with these brush seals, as well as of other types, that as a consequence of the shaft rotation the bristles of the brush seal are exposed to a rotating turbulent flow originating in the higher pressure chamber, which upsets the form and the geometric arrangement of the brush and thus reduces the sealing effect. In particular, the turbulent flow disturbs the exact position of the free ends of the bristles as they project beyond the end edges of circumferential guide flanges towards the rotor or shaft surface. A reliable, leakproof primary seal is not ensured. In addition, particular surface geometries and extensions on the shaft (e.g. screws, steps or indentations) can increase turbulence.

The object of the invention is to provide a brush seal which improves the maintenance of the form and geometry of the bristles for optimum sealing, despite turbulence generated by or at the shaft in the higher pressure chamber in front of the seal.

According to the invention there is provided a brush seal for turbo machines for sealing chambers with

different pressures at a circumferential gap, particularly between a machine housing and a machine shaft, in which a holder for a bristle bundle is provided on the machine housing, from which holder the bristle bundle is fed between circumferential flanges in such a way as to seal against the machine rotor, the bristle bundle forming an axial gap in the circumferential direction opposite the flange facing towards the high pressure side, wherein a cover ring for the bristle bundle is arranged in the axial gap in an axially and radially movable manner, which is activated by the differential pressure and is slidable against the bristle bundle; and

the cover ring protrudes from the axial gap of the ends of the flanges, which surround the machine shaft at a radial distance and encloses the machine shaft with a radial sealing gap.

The cover ring effectively protects the bristle bundle from turbulence formed in the higher pressure chamber. Significant turbulence can be generated for instance by a screwed joint rotating with the shaft and arranged upstream in the higher pressure chamber, the screwed joint having screw heads projecting axially from the shaft. The radially projecting part of the cover ring protects the section of the bristle bundle chiefly responsible for the primary seal at the shaft, this being particularly affected by the formation of turbulence in the fluid. The fluid may be compressed air.

Moreover the primary sealing effect is further increased by the part of the cover ring radially projecting beyond between the end edges of the flanges, especially when the shaft assumes an eccentric position.

The cover ring is pushed axially against the high-pressure side of the bristle bundle by means of the

difference in pressure and the bristle bundle, in turn, is pushed against the inner surface of the flange on the low pressure side to produce a seal. In this way a ring-shaped gap X is formed between the cover ring and the inner flank of the upstream flange, this ring-shaped gap X defining not only a maximum allowable axial travel of the cover ring, but also defining a secondary leakage flow passage and ensuring a uniform pressure distribution around the circumference to provide the pressure on the bristle bundle.

The radial displaceability of the cover ring is dependent on the axial contact pressure on the bristle bundle, which in turn is generated by the pressure differential. In order to keep the axial contact pressure of the cover ring within predetermined limits, fluid adjustment or relief means are preferably provided on the cover ring, such as perforations, through holes or bores in the cover ring. The perforations can be pin-sized, superfine bores, and they can have a sieve-hole type configuration. The term "perforation" in the invention is to be taken also to cover the case where the cover ring is axially porous, enabling flow through the ring.

Preferably the fluid measures (perforations, holes, or bores) are not present in the projecting part of the ring, so that the end part of the bristle bundle projecting beyond the ends of the flange towards the shaft is protected against turbulence as much as possible by a closed, continuous circumferential section of the cover ring, thereby achieving an increased primary sealing effect.

Advantageously, at high shaft eccentricities the cover ring is displaced by the machine shaft; this results in a much smaller gap, at the diametrically opposite side, between the cover ring and the shaft than between the bristles and the shaft. This reduces

primary leakage at the seal.

Embodiments of the invention will now be described, purely by way of example, by means of the drawings, in which:

Fig. 1 shows an axial section of the brush seal, with partial sections of the machine housing and the machine shaft,

Fig. 2 shows a view along C in Figure 1,

Fig. 3a shows a section of the brush seal with basic elements substantially in accordance with Figures 1 and 2, in the stationary operational condition without shaft eccentricity, and

Fig. 3b shows a section of the brush seal showing the effect of shaft eccentricity (Δr).

Figures 1 and 2 show a brush seal for a turbo machine; the brush seal seals a first gap between a machine housing 1 (the stator) and the circumferential surface of a circular machine shaft 2 (the rotor) which is rotatably mounted in the housing between chambers R1, R2 at different fluid pressures. The fluid pressure P1 in the chamber R1 is higher than the fluid pressure P2 in the chamber R2. The primary leakage flow at the seal is shown by the arrow F.

The brush seal includes an annular holder 3 for a bristle bundle B, provided on the machine housing 1 retained between two flange parts 4,5 which together form a circumferential housing for the bristles of U-shaped section, the radially inwardly facing neck of the U being somewhat narrowed. The bristle bundle B emerges from the holder 3 and then extends generally radially inwardly between the essentially parallel first and second flanges 4, 5, which are perpendicular to the rotor surface, in order to form a seal with respect to that surface. The first flange 4 faces the

lower pressure chamber R2 whilst the second flange 5 faces the higher pressure chamber R1. The bristle bundle B is in contact with the inner surface of the first flange 4, but spaced away by a second gap S along the axis from the inner surface of the second flange 5.

The bristles of the bundle B are shown schematically in Figure 2. The bristles are arranged inclined at an angle of 45° to the circumference of the machine shaft 2 and face the rotational direction Dr. They can be in contact with the shaft surface, or, as shown in Figure 1, can be arranged so as to leave a minimum radial sealing gap to the surface of the shaft.

The bristle bundle B is fixed in the machine housing 1 by the holder 3. One radial end of the bristle bundle B is held by a clamping tube 8 substantially in a U shape curled around an annular core 7. The clamping tube 8 has a circumferential slot out of which, radially inwardly, protrudes the bristle bundle B. The clamping tube 8 is fixedly mounted inside an annular ring chamber 9, which is open on one side and formed from two housing parts. The housing parts include the first and second flanges 4, 5. The housing parts are clamped and held in a circumferential groove the base of which is defined by a region 10 of the machine housing 1 with a stepped larger inner diameter, and the walls of which are formed by the step and by the axial end surface of a ring component 11; the latter is screwed to the machine housing 1 along the lines L, L' to retain the housing parts.

All the bristles of the bristle bundle B can be made from a ceramic material, particularly silicon carbide. Alternatively the use of highly-alloyed metallic bristle materials is possible.

In accordance with Figures 1 and 2 a cover ring 6 for the bristle bundle B is arranged so as to be axially and radially movable in the second gap S. The

prevailing differential pressure P1 > P2 acts on the cover ring 6, so that when there is an adequate pressure difference the cover ring 6 is urged axially to the left against the bristle bundle B. The low-pressure side of the bristle bundle B is thus pushed against the inner surface of the first flange 4. The flanges 4, 5 which surround the machine shaft 2 are spaced away from the shaft at equal distances H. The cover ring 6 protrudes radially inwardly from the second gap S between the ends of the flanges and surrounds the shaft 2 leaving a sealing gap or spacing D, being the difference in radius between the shaft and the inner edge of the cover ring. The distance H of the flange from the surface of the machine shaft 2 is chosen to be sufficiently large that even with a relatively large shaft eccentricity Δr (Fig. 3b) the machine shaft does not touch the ends of the flanges 4, 5. In the embodiment described the urging of cover ring 6 by the differential pressure P1 > P2 against the bristles causes it to be spaced away from the inner surface of the second flange 5, forming a ring-shaped gap X, as shown on the right in Figure 1. The width of the second gap S relative to the thickness of the cover ring 6 defines in this way the maximum axial freedom of movement of the cover ring in the second gap S.

The gap X (the ring-shaped gap) is a secondary leakage passage for a relatively small proportion of the fluid fed from the chamber R1 at the higher pressure P1. The fluid flows over the cover ring 6 via a part of the second gap S and from there diffuses through the bristle bundle B into the lower pressure chamber R2.

The reference letter W schematically denotes turbulence in the chamber R1 from which the seal is supplied with a fluid, e.g. compressed air, in accordance with arrow P. The arrangement of the cover

ring 6 protects the bristle bundle B from the turbulently flowing compressed air. In particular there is no adverse influence on the intended arrangement and geometry of the end part of the bristle bundle B, overhanging the ends of both flanges 4, 5 and projecting towards the machine shaft 2. This part of the bristle bundle B is responsible for the primary shaft seal.

In a variation of the embodiment of Figures 1 and 2 the cover ring 6, as shown in Figures 3a and 3b, has identically-shaped through holes or bores 12 distributed around the circumference of the cover ring 6. With a given differential pressure the lateral contact pressure of the cover ring 6 on the bristle bundle B is thus kept within predetermined limits.

Figures 3a and 3b show simple schematic representations in which the position of the machine housing 1 is unchanged while the shaft adopts central and eccentric positions respectively.

Figure 3a shows the stationary operational condition, e.g. of a gas turbine engine. Figure 3b, in comparison, shows the compensation of a comparatively significant shaft eccentricity Δr in operation; at the top the bristles are displaced by the eccentric movement Δr. The maximum gap y from the bristle ends to the shaft occurs at the diametrically opposite location shown at the bottom of Figure 3b. The cover ring 6 also follows the shaft movement. It can be seen that the maximum spacing 2D from the cover ring to the shaft is clearly smaller, i.e. by around more than a half, than the maximum gap y from the shaft to the bristle ends. Consequently in this case, the cover ring 6 predominantly provides the primary seal at the machine shaft 2 which results in comparatively small seal leakage.

The inner diameter of the cover ring 6 relative to

the surface of the machine shaft determines the radial sealing gap D in the normal or stationary operational condition (Figures 1 and 3a), which is constant around the circumference. In the case of Figure 3b this gap has become zero at the top and has increased to twice (2D) its normal width at the bottom.

It will be understood that the invention can also be applied to sealing arrangements where the brush seal is on the shaft facing outwards.

There has thus been described a brush seal for sealing chambers at different pressures at a circumferential gap between a machine housing 1 and the shaft 2. The seal consists of a bristle bundle B mounted in a holder 3 from which it is guided radially inwardly between circumferential flanges 4,5 in order to seal against the shaft. In order to protect the bristles from turbulent flow a cover ring 6 for the bristle bundle is arranged radially and axially movably in the gap between the bristles and the flange 5 on the high-pressure side. The differential pressure urges the cover ring against the bristle bundle. The cover ring projects beyond the inner edges of the flanges and encloses the machine shaft, and in extreme eccentric positions of the shaft (Fig.3b) reduces the maximum gap to twice the clearance of the ring from the shaft.

Claims:

1. A brush seal for sealing the circumferential gap between a shaft and a housing surrounding the shaft so as to separate a higher pressure chamber from an axially adjacent lower pressure changer, in which there is provided:

a circumferential support flange,

a circumferential bristle bundle on a high-pressure side of the flange,

and a cover ring located on the high-pressure side of the bundle, the cover ring being radially moveable and slidable against the bristle bundle, the cover ring protruding radially beyond the flange.

2. A brush seal according to claim 1, in which the cover ring is constructed to allow axial flow through it.

3. A brush seal according to claim 2, in which the cover ring has axial through-holes, bores or perforations.

4. A brush seal according to claim 2 or 3, in which the perforations or the through-holes or bores are arranged essentially in the region of the cover ring facing the flange and not the region protruding from the flange.

5. A brush seal according to any preceding claim and further including a second circumferential flange on the high-pressure side so that the bristle bundle is located between the flanges, there being a gap between the bundle and the flange on the high-pressure side, the cover ring being movable in this gap.

6. A brush seal according to claim 5, in which the two flanges form a circumferentially extending mounting for the bristle bundle, the mounting having a generally U-shaped section with an expanded base region and a narrowed neck region, the base region containing a clamping holder for the bundle and the neck region

substantially containing the cover ring.

7. A brush seal as described herein with reference to the accompanying drawings.

8. A shaft assembly including a rotating shaft, a housing surrounding the shaft and a brush seal according to any preceding claim, in which eccentric movement of the shaft can move the cover ring radially, the flange or flanges are attached to the inside of the housing, and the inner circumference of the cover ring encloses the shaft at a distance such that the maximum gap from the cover ring to the shaft is smaller than the maximum gap from the bristle ends to the shaft.

The Patent Office

11

Application No:	GB 9616775.4	**Examiner:**	Robert L Williams
Claims searched:	1-8	**Date of search:**	12 September 1996

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.O): F2B

Int Cl (Ed.6): F16J 15/32

Other: WPI

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
A	US 5,318,309 Wu-Yang Tseng et al	1

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
		E	Patent document published on or after, but with priority date earlier than, the filing date of this application.
&	Member of the same patent family		